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FATIGUE LIFE ANALYSIS AND TESTS FOR THICK-WALLED CYLINDERS INCLUDING EFFECTS OF OVERSTRAIN AND AXIAL GROOVES

J.H. UNDERWOOD
A.P. PARKER

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US ARMY ARMAMENT RESEARCH, DEVELOPMENT AND ENGINEERING CENTER
CLOSE COMBAT ARMAMENTS CENTER
BENÉT LABORATORIES
WATERVLIET, N.Y. 12189-4050

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J.H. Underwood and A.P. Parker (University of Northumbria at Newcastle, UK)

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13. ABSTRACT *(Maximum 200 words)*
A fracture mechanics-based fatigue life analysis was developed for pressurized thick-walled cylinders autofrettaged by overstrain and with one or several semi-elliptical-shaped axial grooves at the inner diameter. The fatigue life for a crack initiating at the root of the groove was calculated for various cylinder, groove, and crack configurations and for different material yielding conditions. Comparisons were made with fatigue crack growth and laboratory life results from A723 thick-walled cylinders in which cannon firing tests were first performed to produce axial erosion grooves, followed by cyclic hydraulic pressurization to failure in the laboratory. The life analysis, with an initial crack size based on the expected pre-existing defects, gave a good description of the crack growth and fatigue life of the tests for cylinders with and without grooves. General fatigue life calculations summarized important and configurational effects on the fatigue life design of overstrained cylinders, including effects of material yield strength, cylinder diameter ratio, stress concentration factor, and initial crack size.

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INTRODUCTION

The proven method for determining the fatigue life of a pressurized thick-walled cylinder, particularly one with any degree of configurational or loading complexity, is a full-scale test. However, this involves considerable time and cost, so there is always interest in analytical descriptions of fatigue life. The present authors have investigated some aspects of fatigue life assessment for pressure vessels. A comparison of fatigue lives from measurements and analysis showed good correspondence for inner radius-initiated cracking of overstrained cylinders (ref 1), and lifetimes for notched cannon components were well described (ref 2). Recently, the combined problem of fatigue cracking of a pressurized, overstrained thick-walled cylinder with an axial groove at the inner radius has been addressed (ref 3). As expected, the presence of a stress concentration at the critical inner radius location predicted a significant decrease in fatigue life.

The purpose of this investigation is to proceed further with the combined problem of cyclic pressurization of an autofrettaged and axially grooved cylinder. Recent hydraulic pressure fatigue tests of sections of cannons, some of which contained axial grooves from previous firing, are reviewed, and the results are compared with fracture mechanics descriptions of crack growth and fatigue life. Specific comparisons are made between experiment and analysis for the grooved and ungrooved conditions of the recent tests. Then, using analysis that has been shown to give a good description of the fatigue test results, fatigue life predictions are made for various cylinder, groove, and crack configurations and for various applied pressure and material yield strength conditions.

CYLINDER AND TEST CONDITIONS

Figure 1 and Table 1 summarize key features of some recently completed fatigue tests of thick-walled cylinder sections taken from cannon tubes. These tests show one application of the work here on cylinders with axial grooves; any other design requirement or service condition that results in a groove-like stress riser at the cylinder bore would relate to this work. In these tests a section about 1 m long is cut from a cannon and cyclically pressurized from near zero to 670 MPa using procedures summarized in prior work (ref 4). The tube material was a forged ASTM A723 steel of the type commonly used for cannon and other pressure vessels. Nine tube sections were tested, five of which contained significant axial grooves caused by erosion associated with firing. The typical location of the grooves (and the fatigue failure) was in the larger wall ratio portion of the tube section, with outer-to-inner diameter ratio, OD/ID = W = 2.25, see Figure 1. The four remaining tube sections, with no significant grooves, failed in the portion with W = 1.97. Table 1 lists the measured yield strengths of the tube material, S_Y, the tube and groove configurations, and the measured fatigue lives for each of the tests determined as the total number of laboratory pressure cycles required for crack growth completely through to the OD surface. The mean life of the grooved tubes was 3,159 cycles, about 43 percent below that of the tubes with no grooves. This amount of decrease in fatigue life could be significant for many applications. In addition, note that the grooved tubes had the higher wall ratio, which would result in a longer life (if there were no grooves present), so the actual effect of the grooves on life was more significant than the 43 percent reduction noted. This is addressed and discussed in more detail in upcoming sections.

Table 1. Specimen Material, Configuration, and Life Information

Specimen	Yield Strength S_Y MPa	Tube Size at Failure Location OD mm	ID mm	Groove Depth d mm	Measured Fatigue Life N_{meas} Cycles
#3	1120	270	120	0-5	2,764
#5	1030	270	120	0-6	3,390
#9	1060	270	120	0-5	3,441
#23	1060	270	120	0-7	2,894
#49	1090	270	120	0-9	3,306
					Mean: 3,159
#6	1110	309	157	0	5,040
#14	1140	309	157	0	5,365
#21	1110	309	157	0	5,360
#85	1120	309	157	0	6,593
					Mean: 5,590

Another aspect of the tests that should be characterized is the determination of a typical size and spacing of grooves for the affected tubes. These features showed considerable variation, but after repeated comparisons and measurements, an idealized groove configuration was identified that was typical of the five tubes as a group. It is depth, d, of 4 mm, width of 16 mm, and space between grooves of 25 mm, as shown in Figure 2. The length of the grooves down the axis of the tube is typically a few hundred millimeters. This characterization of the grooves was used in the upcoming fracture mechanics and fatigue analyses.

GROOVED CYLINDER ANALYSIS

Before a fatigue life analysis can be performed for a crack growing from a groove in a cylinder, the notch-tip stress and stress intensity factor must be determined. This is accomplished by a superposition of stresses in a manner similar to that of other recent work (refs 3,5). Referring to Figure 3, the stress of interest is the circumferencial stress at the tip of a groove in a pressurized cylinder, called S_P. It is calculated by elastic superposition of two other stresses that are easier to obtain: the stress, S_1, at the same location in an identical cylinder loaded by external tension of the same magnitude as the pressure; and the stress, S_2, in a cylinder under hydrostatic pressure. It can be seen that the external loads cancel one another, so that $S_P = S_1 + S_2$. Then S_P can be determined from the stress S_1 for the cylinder under external tension, where, because there is no effect of pressure on the groove surfaces, the stress at the groove tip

$$S_1 = -p\, k_t[2W^2/(W^2-1)] \tag{1}$$

is simply the Lamé stress for a cylinder under external tension (ref 6), -p $[2W^2/(W^2 - 1)]$, times the stress concentration factor for the particular configuration of the groove, k_t. The convention here is that p is negative. Using Eq. (1) and the superposition, the circumferential stress at the tip of groove, S_p, is

$$S_p = -p[(2k_t - 1)W^2 + 1]/[W^2 - 1] \tag{2}$$

Note that in the limit as $k_t \rightarrow 1$, Eq. (2) reduces to the Lamé stress at the ID of a cylinder with internal pressure (ref 6), as it should. Equation (2) accounts for both the pressure on the ID surface of the cylinder and the pressure on the groove surfaces.

Finally, the early work of Inglis (ref 7) was used to determine the value of k_t as a function of the depth to half width ratio, d/c, of the groove and the spacing of the grooves relative to half width, b/c, as follows,

$$k_t = (1 + 2d/c) fn(b/c) \tag{3}$$

where *fn*(b/c) reduces the k_t calculated from the Inglis solution to account for the effect of multiple, closely spaced grooves. An *fn*(b/c) = 0.63 was obtained from a solution for multiple, closely spaced holes in the well-known Peterson k_t compendium (ref 8) for the b/c = 3.13 of the typical groove configuration of Figure 2. For the d/c = 0.5 from Figure 2, the resultant k_t is 1.26. Equation (2) and this result from Eq. (3) were used next in expressions for stress intensity factor, K, and fatigue life, N.

K AND FATIGUE LIFE ANALYSIS

A simple, short-edge-crack analysis of K and fatigue life was used, so as to not lose focus on the effects of the groove and also because experience has shown that the dominant control of fatigue life is from short cracks. Starting with the experimentally-determined relationship describing the fatigue crack growth rate in terms of the stress intensity range, ΔK (ref 1), we have

$$da/dN = 6.52 \times 10^{-12} \Delta K^3 \tag{4}$$

where 6.52×10^{-12} and the power three describe the growth rate for the A723 steel used and are appropriate for da/dN in m/cycle and ΔK in MPa $m^{1/2}$. The definition of ΔK is $\Delta K = K_{max} - K_{min}$ for $K_{min} > 0$ and $\Delta K = K_{max}$ for $K_{min} < 0$, as in prior work (ref 1). ΔK is determined using the very familiar expression for short cracks, as follows,

$$\Delta K = 1.12 h (\pi a)^{1/2} S_{eff} \tag{5}$$

The use of a short-crack expression for ΔK accounts for the stresses at the groove surface but does not account for the change in stress as the crack grows away from the groove. In Eq. (5), a

crack shape factor, h, has a value of one for the straight-fronted cracks of primary concern here, and S_{eff} is a combined stress that includes the three types of loading of importance in a pressurized cylinder with a pressurized crack, described as follows.

The expression for S_{eff} is

$$S_{eff} = S_P + S_R - p \tag{6}$$

where S_P is the concentrated circumferencial stress at the groove tip and includes the effects of the k_t of the groove and the pressure on both the groove surfaces and the ID surface. It has already been described in Eq. (2). The stress S_R is the concentrated circumferential stress at the notch tip due to the autofrettage residual stress in the cylinder and can be obtained from

$$\begin{aligned} S_R &= S_Y k_t [1 - \ln W\{2W^2/(W^2-1)\}] \qquad & S_R \le S_Y \\ S_R' &= S_Y & S_R > S_Y \end{aligned} \tag{7}$$

The top expression is, apart from k_t, the usual expression for circumferencial residual stress at the ID of a fully overstrained cylinder (ref 9), that is, one in which yielding has proceeded throughout the wall thickness. Including k_t in the expression for residual stress accounts for the effect of the groove. However, since the unconcentrated residual stress often approaches the yield strength and k_t can be well above one, the requirement in the lower expression has been added. This limits S_R to the material yield strength, which is believed to occur in reality as a result of yielding at the tip of the groove. Equation (7) accounts for the effect of yielding on S_R but does not account for any strain hardening that may occur. Finally, the inclusion of the value of the pressure, p, in S_{eff} accounts for the effect of pressure on the crack faces. The rationale for this is the fact that the K for a given crack configuration with a pressure applied to the crack faces is exactly equivalent to the K for the same crack with remotely applied tension of the same magnitude, that is, with t = - p (ref 1).

The expression for fatigue life, N, can thus be written by integrating Eq. (4) and combining the result with Eqs. (5) and (6) to give

$$N = 2[1/\sqrt{a_i} - 1/\sqrt{a_f}]/6.52 \times 10^{-12}[1.12\sqrt{\pi}\, h(S_P + S_R - p)]^3 \tag{8}$$

where the term $2[1/\sqrt{a_i} - 1/\sqrt{a_f}]$ is from integration and the other terms are from Eqs. (4) through (6). In combination with Eqs. (2), (3), and (7), this expression can be used to calculate fatigue life taking into account the key material, configuration, and loading factors which control fatigue life in a pressurized, grooved, overstrained thick-walled cylinder. These factors, referring also to terms and parameters in the above equations, are initial and final crack depths (a_i, a_f); material crack growth properties (6.52×10^{-10} and power three); the edge crack nature of the overall configuration ($1.12\sqrt{\pi}$); the semi-elliptical or straight shape of the crack (h); pressure on the ID and groove surfaces (S_P); the overstrain residual stress (S_R); pressure in the crack (p in Eq. (8)); diameter ratio (W in Eqs. (2) and (7)); stress concentration of the groove (k_t in Eqs. (2) and (7)); material yield strength (S_Y in Eq. (7)).

Fatigue life calculations were performed based on Eq. (8), using just one arbitrarily selected input value, that of the initial crack depth, a_i. All other inputs were based on material and test conditions, and a_i was selected with at least some consideration of test conditions. Relatively few firings (100 to 200) of the type expected to initiate heat check-type cracks had occurred with the tubes tested here, so a relatively small a_i of 0.1 mm was used for all calculations. The a_f was taken as the wall thickness, but this is known to have much less effect on the N than a_i does. The comparison of the life calculations with experimental results is described next.

CALCULATED AND MEASURED LIVES

Table 2 lists the values of residual stress, S_R, used for the calculated life of each of the nine tubes. Note that the residual stress was limited to the value of the yield strength for each of the five grooved tubes. Calculated lives for the W and k_t values of the tests are listed, and ratios of calculated to measured life are also listed, with the direct comparisons between calculated and measured life indicated by an asterisk (*). Referring to the life ratio results, it is clear that the calculations give a good description of measured life for both grooved and ungrooved cylinders. Furthermore, if no grooves had been present in the W = 2.25 portion of the cylinder and a life had been measured there, it would have been four to eight times higher than that with the grooves. Conversely, if there had been grooves with a k_t of 1.26 in the W = 1.97 portion of the tube, the life would have been about four times lower than that without the grooves. So it is clear that the grooves, even with a k_t as low as 1.26, have a significant effect on life.

A comparison of calculated and measured crack growth behavior is shown in the plots of Figure 4 for two of the tests--one with grooves (and k_t = 1.26) and one with no grooves. Crack depth was measured at various points during the tests using a pulse-echo ultrasonic method from the outside surface of the cylinder. Calculations of crack depth versus number of cycles were obtained from Eq. (8) by simply setting a_f equal to the desired crack depths. The reasonably good agreement in placement and shape of the curves can be seen. One feature noted with the measured curves, and also to a lesser extent with the calculated curves, is the higher growth rate at small crack depths of the grooved specimen compared with the rate of the ungrooved specimen. The higher rate is apparent for crack depths up to about 4 mm, the typical depth of the groove, so St. Venant's principle of local effects being displayed near a discontinuity is once again seen. The expected increase in crack growth rate, da/dN, can be approximated using a generic expression for growth rate, as follows

$$da/dN = constant(k_t \Delta S a^{1/2})^n \tag{9}$$

and since only k_t is different for a grooved versus an ungrooved sample, an expression for the ratio of growth rates can be written as

$$(da/dN)_{GROOVE}/(da/dN)_{NO\ GROOVE} = (1.26/1.00)^3 \approx 2.0 \tag{10}$$

The results in Figure 4 are generally consistent with the predictions of Eqs. (9) and (10).

Table 2. Comparison of Calculated and Measured Fatigue Lives

Specimen	Residual Stress S_R MPa	Calculated Life N_{calc} Cycles		Life Ratio N_{calc}/N_{meas}	
		$k_t = 1.00$	$k_t = 1.26$	$k_t = 1.00$	$k_t = 1.26$
W = 2.25:					
#3	-1120 (S_Y)	22,720	3,964	8.22	1.43*
#5	-1030 (S_Y)	14,418	3,049	4.25	0.90*
#9	-1060 (S_Y)	16,652	3,319	4.84	0.96*
#23	-1060 (S_Y)	16,652	3,319	5.75	1.15*
#49	-1090 (S_Y)	19,373	3,623	5.77	1.08*
	Mean:	17,963	3,455		
W = 1.97:					
#6	-916	5,353	1,502	1.06*	0.27
#14	-941	5,831	1,588	1.09*	0.28
#21	-916	5,353	1,502	1.00*	0.27
#85	-924	5,500	1,529	0.83*	0.27
	Mean:	5,509	1,530		

*Direct comparison of measured and calculated life.

A broader comparison between calculated and measured fatigue behavior is shown in Figure 5. The results of the five grooved tests, with 3,159 cycles average life, and the four ungrooved tests, with 5,590 cycles average life, are plotted. Two calculated curves are shown--one for the W = 2.25 of the grooved tests and k_t values of 1 to 2, and one for the W = 1.97 of the ungrooved tests and the same k_t values. The calculations were from Eq. (8) using the appropriate W and k_t values. The agreement between experiment and theory is good, as discussed earlier, and it instills confidence in the use of this type of plot for fatigue life prediction and design. It is clear that both diameter ratio and stress concentration factor have profound effects on the fatigue life of pressurized thick-walled cylinders.

DESIGN CALCULATIONS

The approach used above to make comparisons between measured and calculated lives for a specific series of tests can be used for general calculations that may be of interest in fatigue life design and analysis of pressurized cylinders. Equation (8) can be used as before, taking a broader range of key variables as input to the equation, including wall ratio, stress concentration factor, and initial crack size. Such calculations were made, and results are shown in Figures 6 and 7. One other factor was addressed in the calculations--the effect of limiting the residual stress due to yielding, as discussed earlier in relation to Eq. (7). Lives were calculated with and without the yielding limitation and compared. The calculations were for $S_Y = 1100$ MPa, for the growth rate of Eq. (4), for a pressure of 550 MPa, and for S_R of a fully overstrained cylinder (Eq. 7).

Figure 6 gives results that show the effects of diameter ratio and stress concentration on fatigue life for the same a_i as before, 0.1 mm. Results are given with the limitation of $S_R < -S_Y$ and with no limitation. Note the factor of ten decrease in life due to a moderate decrease in W or a moderate increase in k_t. Note also the significant additional decrease in life associated with the $S_R < -S_Y$ limitation for large W and k_t, conditions that favor yielding and thus shorten life because the yielding eliminates some of the beneficial compressive residual stress. Figure 7 shows the effect of initial crack size on fatigue life for the largest W value of Figure 6; the middle curve of each set of three in Figure 7 is the same as the upper curves in Figure 6. As expected from the form of Eq. (8), larger values of a_i result in significantly lower lives. And, again in this comparison, the effect of the $S_R < -S_Y$ limitation is to decrease the calculated fatigue life. These results in Figures 6 and 7 provide at least a capsule summary of important material and configurational effects on the fatigue life of overstrained cylinders, including effects of material yield strength, the cylinder diameter ratio, the stress concentration factor, and the initial crack depth.

SUMMARY

1. Elastic superposition was used to calculate the concentrated stress at the root of a semi-elliptical groove at the inner diameter of a pressurized, overstrained cylinder. This stress was combined with the residual stress due to overstrain and the stress due to pressure in the crack to develop fracture mechanics calculations of fatigue life.

2. Calculated fatigue life compared well with measured life from A723 steel cylinder tests with two different diameter ratios and stress concentration factors and using an initial crack size based on the expected pre-existing defects in the cylinder.

3. Calculated effects on fatigue life were summarized for use in fatigue design and analysis of overstrained cylinders. Significant decreases in calculated life are expected due to: (1) the limitation of the overstrain residual stress to a value equal to the yield strength; (2) a decrease in the cylinder diameter ratio; (3) an increase in the stress concentration factor; and (4) an increase in the initial crack size.

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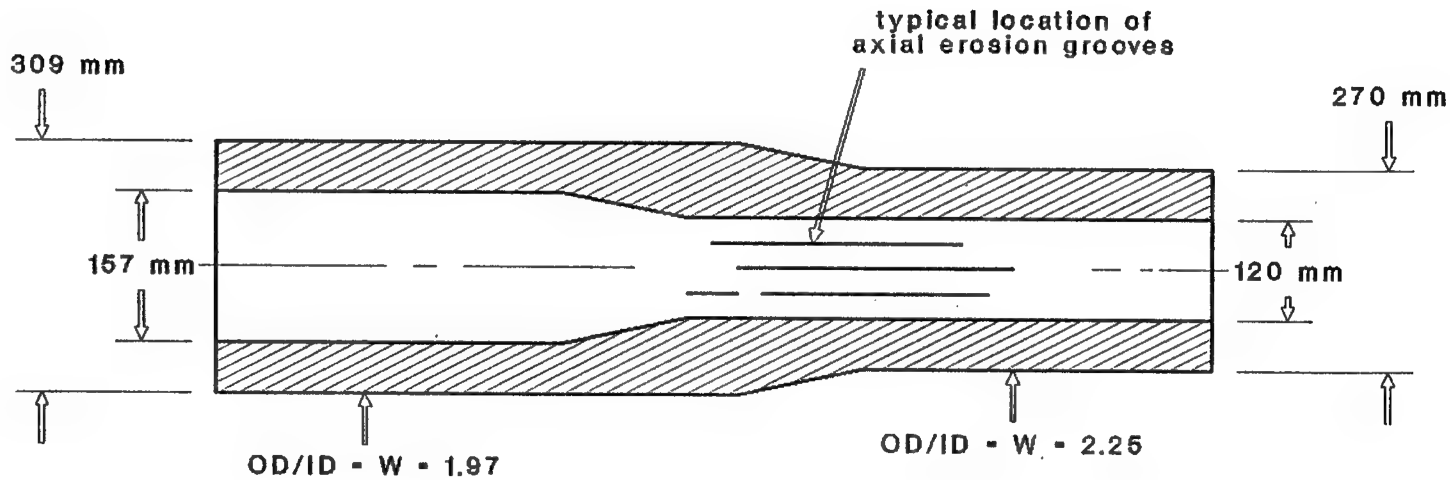

FIG. 1 - Specimen Configuration

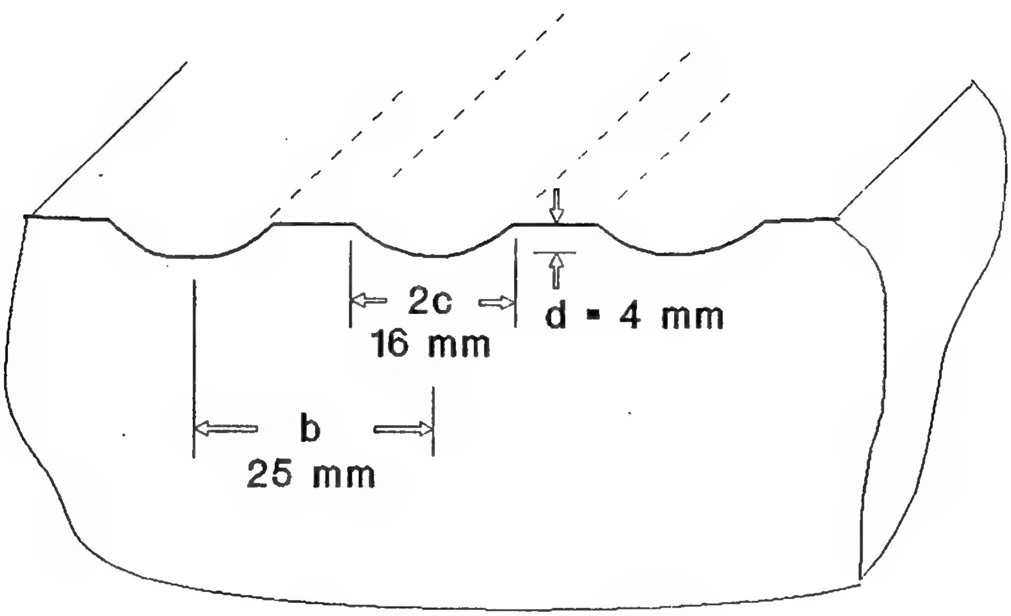

FIG. 2 - Idealized Groove Configuration

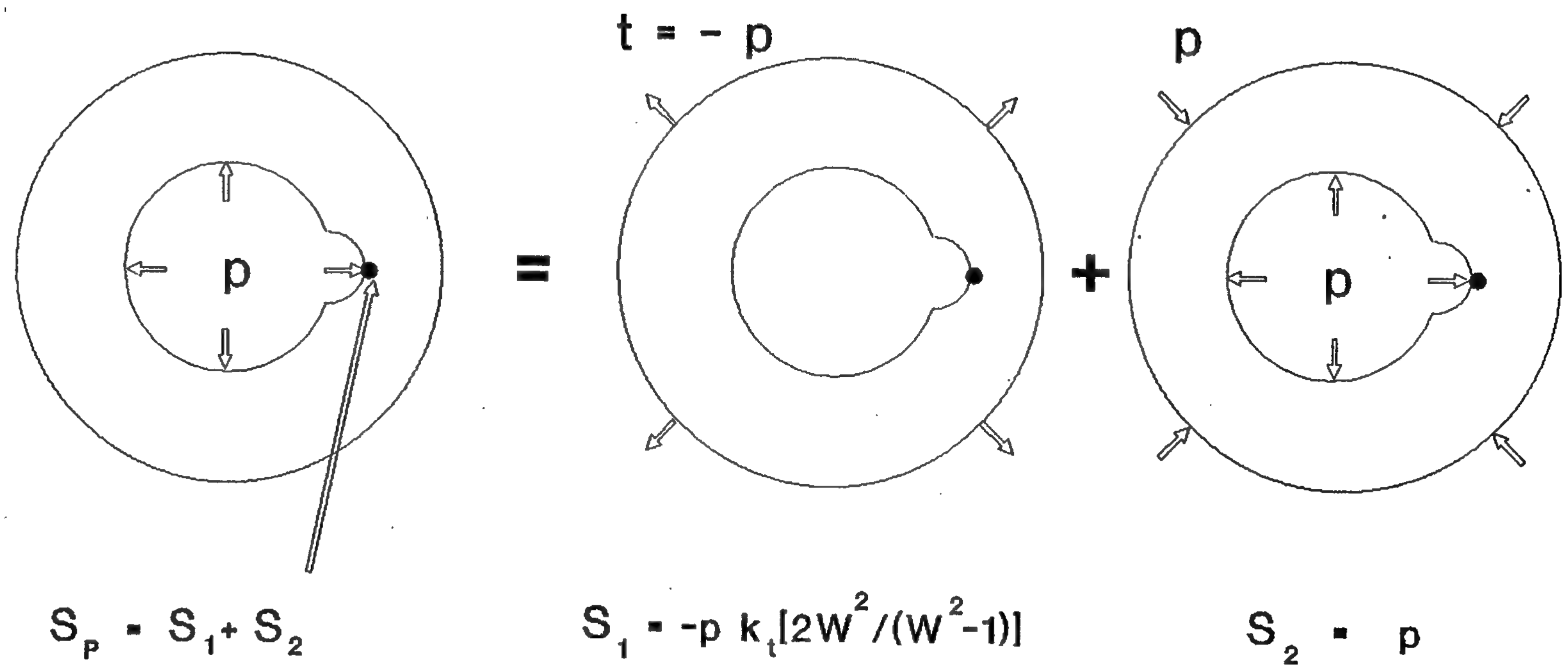

FIG. 3 - Calculation of Notch-Tip Stress by Superposition

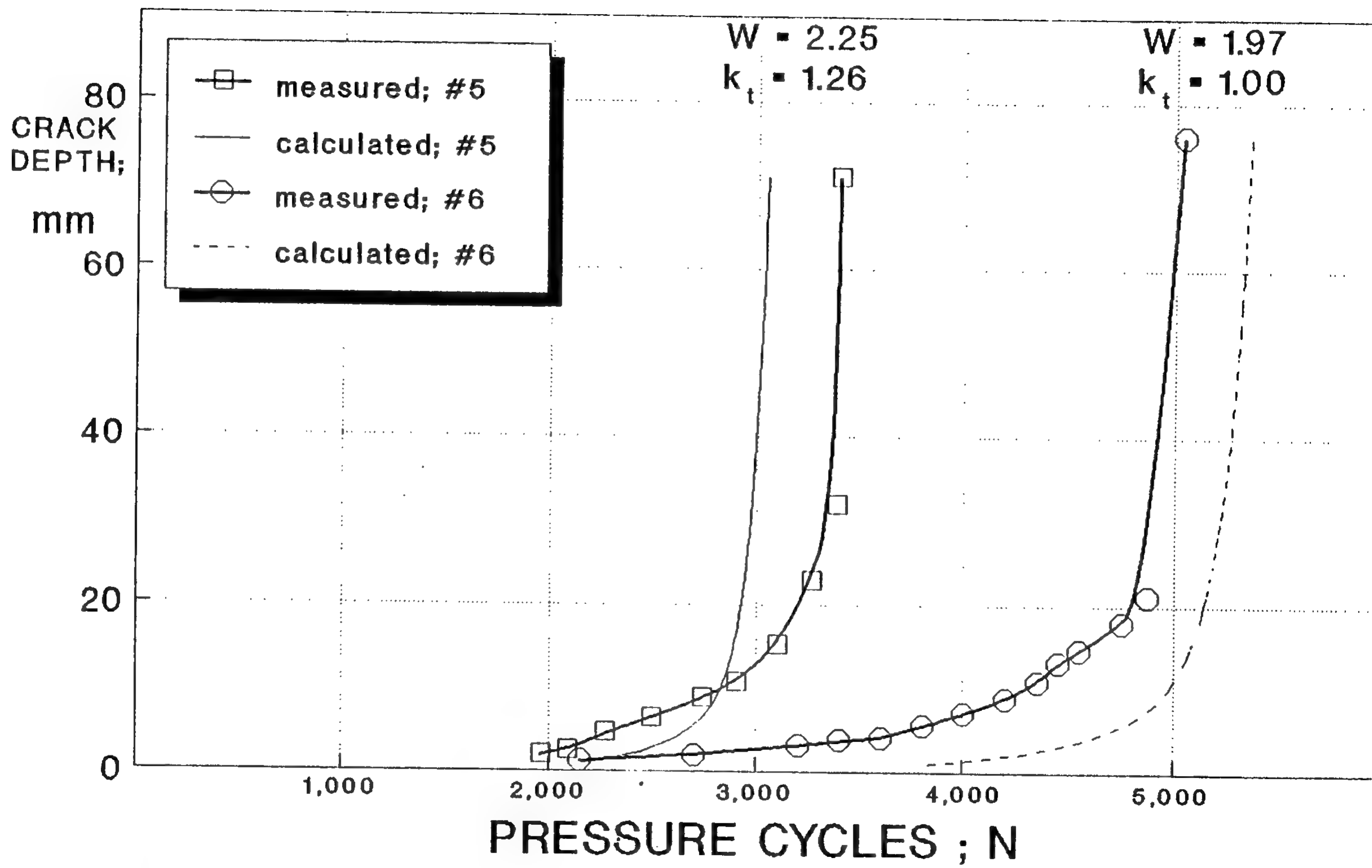

FIG. 4 - Measured and Calculated Crack Growth Versus Number of Fatigue Cycles

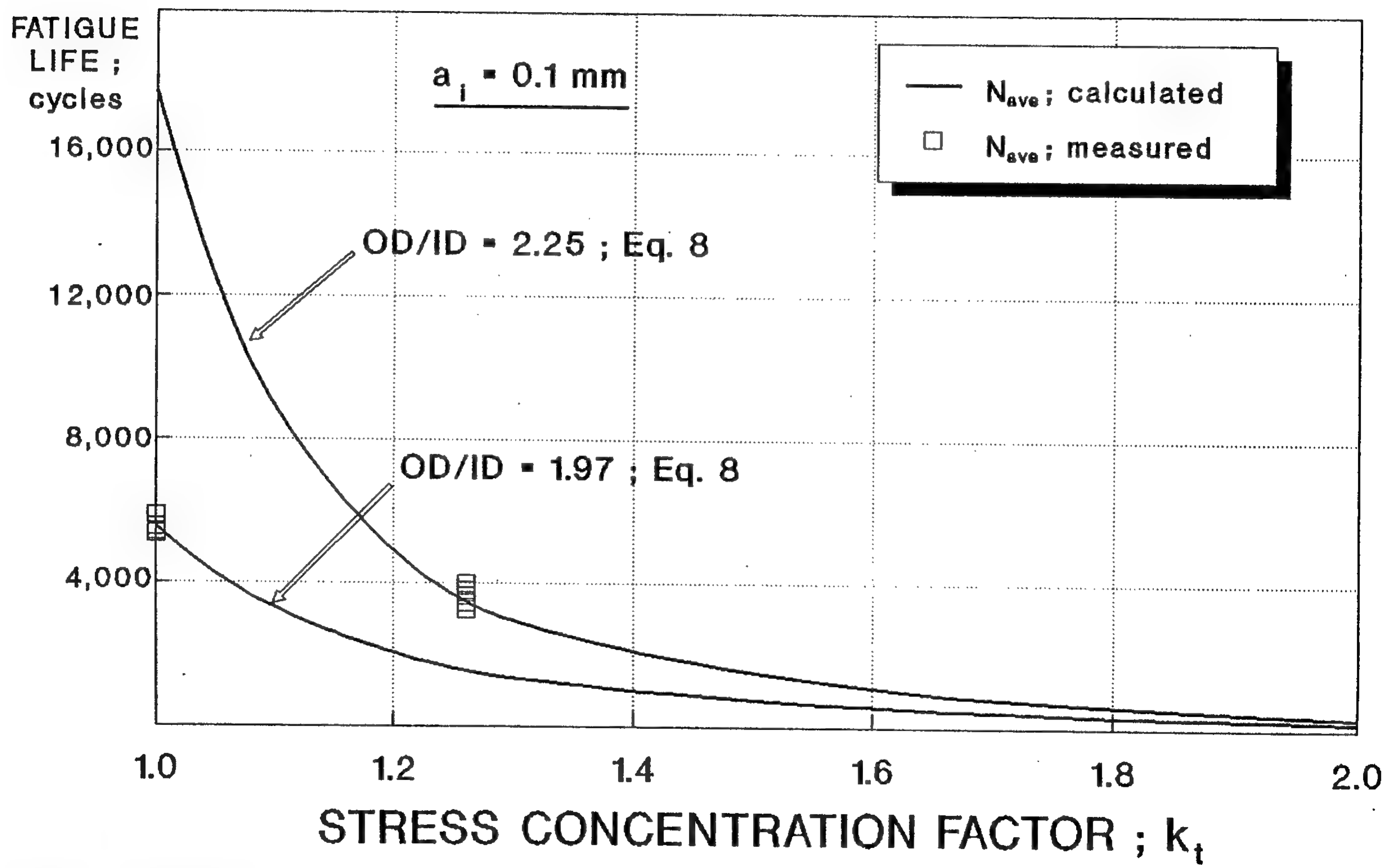

FIG. 5 - Effect of Stress Concentration Factor, k_t, on Calculated Fatigue Life

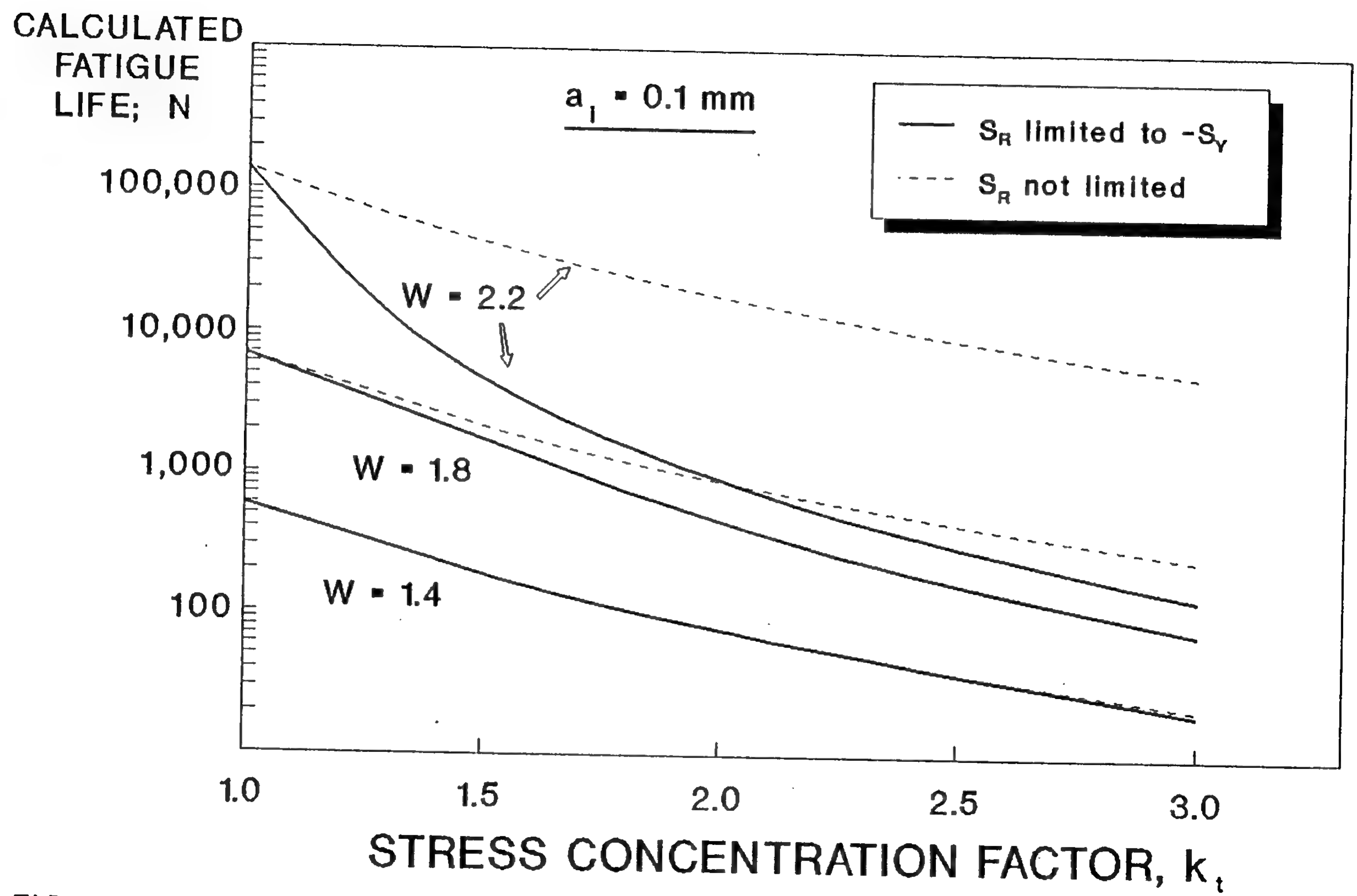

FIG. 6 - Effect of Wall Ratio, Stress Concentration and Yielding on Life

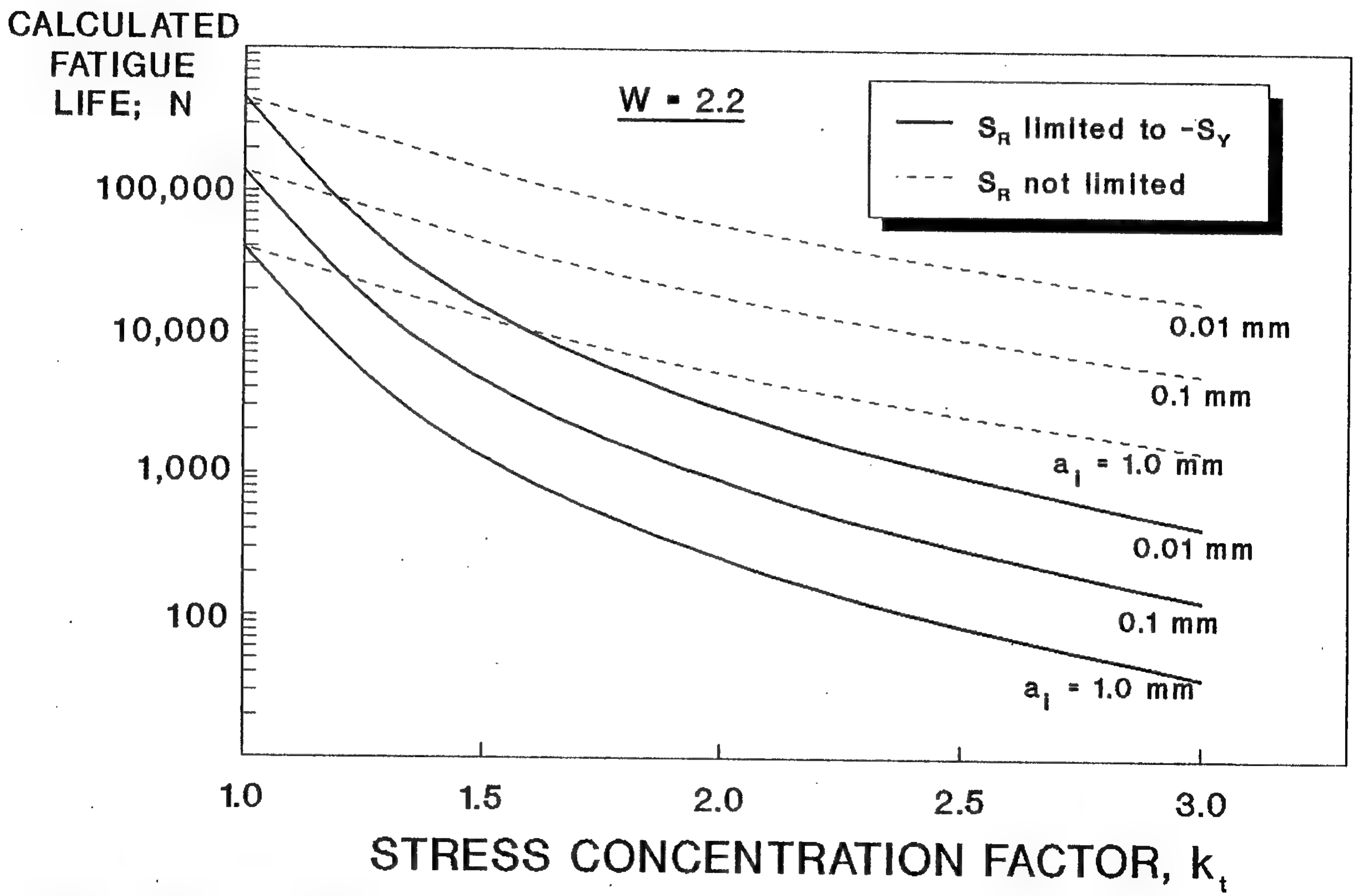

FIG. 7 - Effect of Stress Concentration, Yielding and Initial Crack Size on Life

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